REMARKS

Claims 11-20 are pending in this case. Applicants herein amend claims 11, 13 and 17, cancel claim 20, and add new claims 21-24. The amendment to independent claim 17 incorporates the elements of allowable claim 20, the latter now cancelled. Claim 21 is old claim 11 adding limitations from claims 12 and 15. No new matter is added.

After entry of these amendments, claims 11-19 and 21-24 stand pending in this case and are presented for examination. Applicants respectfully request allowance of the present application in view of the foregoing amendments and the discussion herein.

Claim Rejections Under 35 USC 112

Claims 11-16 stand rejected under 35 USC 112, second paragraph, as being allegedly indefinite for failing to particularly point out and distinctly claim the subject matter which applicant regards as the invention.

Applicants have amended claim 11 to conform with the suggestion of the Examiner in the Office action. Reconsideration and withdrawal of this claim rejection are respectfully requested.

Claim Rejections Under 35 USC 102

Claims 11-14 and 17-18 stand rejected under 35 USC 102(b) as being allegedly anticipated, separately, by each of the following three references (with discussion including arguments following each reference's identification.

WO 03/023221: The Office action alleges that this reference anticipates claim 11, essentially stating that this reference recites the text of claim 11. It is noted that WO 03/023221 teaches a constant supplying of the high pressure pump associated with an intermittent and ongoing bypass shunting downstream of the high pressure pump under low load conditions (see, for example, page 5, lines 10-22, which also indicates an ongoing fluctuation in pressure, rather than a stable reference pressure). Also, although the Office action states "the high-pressure pump is supplied with an adjustable fuel flow via valve 36; adjusting the pressure to the reference pressure in a first operating mode during high speed operation by regulating the fuel flow of the fuel delivered to the high-pressure pump . . .", 36 is the mass control valve that is in

the bypass part of the embodiment depicted in FIG. 1 of this reference. This appears to be an error. There is a flow control orifice 38 immediately upstream of the high-pressure pump 18. However, this is not shown to be under control of the controller/ECU, and any control of maximum flow by such flow control orifice 38 would appear to be based on presetting to a desired open position. This teaches away from the present invention as claimed in claim 11 because claim 11 teaches in part, ". . . adjusting from a control unit that is connected to a volume flow control valve upstream of the high-pressure pump the pressure in the fuel pressure accumulator in a first operating mode to the reference pressure by regulating the fuel flow of the fuel delivered to the high-pressure pump depending on the fuel volume which must be injected and the reference pressure . . ."

Overall, this reference teaches a different approach to controlling fuel volumes in an accumulator. Based on the above distinctions, individually or in any combination, and/or the amendment of claim 17 herein, it is respectfully asserted that this reference does not anticipate claim 17.

Reconsideration and withdrawal of this basis of rejection is respectfully requested.

DE 196 12 413: Based on the automatic English translation (copy provided herein) of this reference provided at the European Patent Office website www.espacenet.com, the object of the pressure relief valves (50 in FIG. 1, 150 in FIG. 2) positioned along a bypass line (44 in FIG. 1) is to guarantee "a fast and reliable pressure relief of the common rail" (14 in FIG. 1, 114 in FIG. 2) in case of an emergency (see paragraph beginning "In Fig. pressure . . ."). Also, in the following paragraph it is stated that when the control member 154, positioned between the variable suction throttle/valve 128 (controlled by the ECU 18 via 130) and the pressure relief valve 150, moves "beyond the suction pressure control range," the pressure relief valve opens. It appears that this movement beyond the normal range for 128 is to the closed side of the range, so that when there is no need for additional fuel to the high-pressure pump 116, such as due to an emergency rise in pressure in the accumulator/common rail 114, then the linkages between the variable suction throttle/valve 128 operate to open the pressure relief valve. This, though of value for its intended purpose, does not teach the present invention as claimed. In part there is no provision in this reference to operate to maintain the reference pressure in the second operating mode (and it is believed that the statement in paragraph number 5 of the Office action is accordingly not a correct representation of the teachings of this reference).

Based on the above distinctions, and/or the amendment of claim 17 herein, it is respectfully asserted that this reference does not anticipate claim 17.

Reconsideration and withdrawal of this basis of rejection are respectfully requested.

DE 197 14 489: Based on the automatic English translation (copy provided herein) of this reference provided at the European Patent Office website www.espacenet.com, the reference teaches various specific feather/spring linkages between a volume closing member 20, moved by an actuator 21 (which is controlled by controller/ECU 6), and a pressure closing member 22. Thus, a single actuator 21 controls both the flow between the pumps 2 and 23, going into the reservoir/accumulator 4, and the flow from return pipe 26 to direct flow through volume closing member 20 back to fuel tank 11. There does not appear to be separate adjustings addressing two distinct operating modes as claimed in claim 11 as amended herein. Instead, it appears that there is a control of both valves 20 and 22 by a single actuator.

Further, from the explanation offered in the paragraph beginning "The arrangement after Fig. 1 . . .," there does not appear to be a teaching in this reference of operating with two operating modes as claimed herein, including all independent claims herein.

Based on the above distinctions, and/or the amendment of claim 17 herein, it is respectfully asserted that this reference does not anticipate claim 17.

Accordingly, reconsideration and withdrawal of this basis of rejection are respectfully requested.

(please continue to page 10)

Conclusion

The commissioner is hereby authorized to charge any appropriate fees due in connection with this paper, or credit any overpayments to Deposit Account No. 19-2179.

Respectfully submitted,

Dated: 7/9/07

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Result Page

The invention concerns a pressure fluid utility system, in particular for a fuel injection system for example for a Diesel internal-combustion engine, in accordance with the generic term of the patent claim 1.

Such pressure fluid utility systems become as Common Rail (CR) - systems designates, with which the individual consumers, i.e. with fuel injection systems the individual injection nozzles attached to a common are high pressure pipe fed by the high-pressure pump. In accordance with a well-known version, as it is described for example in the DE 41 26 640 A1, a proportional pressure relief valve is attached to the Common Rail, which has the function to have flowed back in each case as much from the pump promoted fuel that in the COMM on Rail the pressure wished in each case adjusts itself. This pressure relief valve can also the task be transferred to diminish in the case of errors in the system the pressure into the Common Rail fast to a lower value.

With this well-known system the losses of energy are unfavorable due to the pressure fluid and/or fuel diverting over the pressure relief valve. It is more favorable to vary the pressure in the Common Rail by a variable delivery the high-pressure pump. A CR system with this kind of printing control is from the EP 0,299,337 a2 is admitting and forms the generic term of the patent claim 1. The variable delivery of the pressure fluid and thus the adjustment of the pressure in the Common Rail and/or the common high pressure pipe to the pressure fluid need of the consumers upstream suction throttle by one the high-pressure pump, which is adjusted accordingly with help one of a pressure sensor in the Common Rail of seized pressure value, is made that the seized pressure takes the desired value. Also further a pressure relief valve attached to the Common Rail is intended, which serves however only for the maximum pressure delimitation.

With the pressure fluid utility system of the Common Rail design in accordance with the generic term of the patent claim 1 it is unfavorable that if in the system an emergency is determined, for example wedging a Injektionsnadel or if a very fast adjustment to a smaller load and thus the attitude are necessary on a smaller injection pressure, which cannot be diminished pressure in the COMM on Rail fast enough.

The invention is therefore the basis the task to train a pressure fluid utility system further in such a manner in particular for a fuel injection system in accordance with the generic term of the patent claim 1 that at small device-technical expenditure when occurrence emergencies at the system a faster pressure drop is made possible.

This task is solved by the characteristics of the patent claim 1.

According to invention to the Common Rail, which is fed by means of a suctionthrottled pump, a pressure relief valve attached, which is activatable in emergency, i.e. when the occurrence events, which require a pressure drop in extremely short time in the Common Rail, preferably as a function of the pressure or flow conditions within the range of the suction throttle. False demands of the pressure fluid utility system on the one hand and the assigned aggregate like e.g. the engine on the other hand, can be excluded in this way effectively. Due to the activation of the pressure relief valve as a function of Druckund/or to flow conditions within the range of the suction throttle the device and in particular can be kept small the technical circuiting expenditure, in which the control condition of the suction throttle becomes usable the production of a suitable adjusting signal for the pressure relief valve. The organization according to invention of the pressure fluid utility system provides thus for an effective security with statement of an emergency at the system, without an additional and complex high pressure relief valve with magnet manipulation (DBE valve) would have to be planned.

Favourable training further are the subject of the Unteransprüche.

The further training of the patent claim 2 has the advantage that the number of moved parts is reducable, in which the movement of the suction throttle as placing movement for the pressure relief valve is consulted.

A further simplification of the structure results with the further training of the requirement 3, in which the placing element for the suction throttle the task of the manipulation of the pressure relief valve will transfer at the same time.

If the pressure relief valve responds only after exceeding of a border regulating distance of the suction throttle, a increased insurance of operation of the pressure fluid utility system results. It can be guaranteed with these measures reliably that the pressure relief valve is released actually only in emergencies and not within the range of usual placing movements of the suction throttle.

The use of the placing movement of the suction throttle for the activation of the pressure relief valve can be realized particularly simply if the placing element of the suction throttle is formed by a slidegate valve piston in accordance with requirement 6.

It was shown that it is sufficient perfectly for the effective security of the CR (Common Rail) system by need-wise fast pressure drop to design the pressure relief valve as poppet valve whose valve body is up impactable in accordance with the favourable further training of the requirement 7 of the placing element of the suction throttle against a resetting force.

In order to be able to limit the power consumption of the drive for the placing element of the suction throttle for the case that the valve body of the pressure relief valve is to be shifted against a very high thrust force of the Common Rail, the further training is in accordance with requirement 10 of special advantage. The kinetic energy of the placing element of the suction throttle is used here for the support of the unblocking of the pressure relief valve. The drive for the placing element of the suction throttle, for example the magnetic field strength can in this way be held relatively small, whereby a sufficient Aufsteuerkraft is made available nevertheless, in which the placing element of the suction throttle with starting distance against the valve body locking high pressure off strikes. Of course it is also possible to consult for taking the valve body off another force translation for example a stroke assistance.

A further possibility of the realization of the activation according to invention of the pressure relief valve is the subject of the requirement 13. In accordance with this further training the pressure relief valve is activated if the flow quantity of the suction throttle falls below a certain value. For the derivative of a suitable adjusting signal for the pressure relief valve here also that can be consulted printing Rome starting from the high-pressure pump, if the pressure relief valve is switched between Common Rail and high-pressure pump.

A favourable arrangement of such a pressure relief valve is the subject of the requirement 15. In favourable way by the throttle effect of the check valve built in the valve body use is made. The valve body locks the Common Rail off until in the mechanical handling operations of the high-pressure pump at the check valve a pressure drop takes place. If the suction throttle closes, then the delivery of the high-pressure pump reaches zero, whereupon the check valve closes and the pressure in the COMM on Rail opens the valve body and makes a connection to the tank.

The arrangement in accordance with requirement 16 has the further advantage that with simple means in the mechanical handling operations of the high-pressure pump a difference of the surfaces loaded by pressure is made available to both sides of the valve body, with which the tumbler of the valve member is supported.

The pressure relief valve can be activated in accordance with the further variant according to requirement 18 also as a function of printing Rome on the suction throttle, which the subject of the requirement 18 is. This variant is characterised in particular then by a good Ansprechcharakteristik, if the suction throttle between a Vorförderpumpe and the high-pressure pump is arranged. If with the occurrence of an emergency situation the Vorförderpumpe is thus switched off, the tax chamber of the pressure relief valve is relieved suddenly, so that the pressure relief valve opens.

It is finally also possible to activate the pressure relief valve in accordance with requirement 19 as a function of printing Rome starting from the suction throttle. By the effect use is made that this pressure drops after the Zusteuern of the suction throttle and during continuous drive movement of the high-pressure pump fast and is thus made available in shortest time a strong adjusting signal for the tax chamber of the pressure relief valve.

The arrangement of the pressure relief valve in accordance with requirement 21 opens the possibility of a favorable force translation.

The dynamics of the relief valve can be still increased with the further training of the requirement 23.

Below on the basis schematic designs several remark examples of the invention are more near described. Show:

Fig. 1 a hydraulic circuit of the pressure fluid utility system in the arrangement as COMM on Rail (CR) - fuel injection system with solenoid operated suction throttle regulation and emergency pressure relief;

Fig. 2 a detail of a further execution form of the pressure fluid utility system in a somewhat modified arrangement of the control of the pressure relief valve;

Fig. 3 a variant of the pressure fluid utility system with a modification of the control of the pressure relief valve;

Fig. 4 a concrete execution form of the control of the pressure relief valve with a mechanical coupling between a placing element of the suction throttle and a tappet for the valve body of the pressure relief valve;

Fig. 5 a further execution form pressure fluid of the utility system with a modified control section for the pressure relief valve;

Fig. 6 the detail VI in Fig. 5;

Fig. 7 a further execution form pressure fluid of the utility system with a modified arrangement and activation of the pressure relief valve; and

Fig. 8 one the Fig. 7 similar cutout of the hydraulic system for the pressure fluid utility system with a modified control of the pressure relief valve.

Fig. 1 shows schematically the hydraulic system for a fuel injection system of a Diesel internal-combustion engine, whereby fuel injection system as ?Common Rail? - system (CR system) is trained. With the reference symbol 10 a mehrzylindrige internal-combustion engine, for example a Diesel internal-combustion engine shown, is a set of fuel injection nozzles 121 to 12n are assigned to which. The injection nozzles are attached to a common high pressure pipe 14, i.e. to a so-called ?Common Rail?, which is fed of a high-pressure pump 16. The high-pressure pump 16 is preferably formed by a radial piston pump. With 17 a memory is designated, which is attached to the Common Rail 14.

With the reference symbol 18 an electronic control unit is designated, with which over the electrical control lines 201 to 20n the injection nozzles 121 to 12n are addressed. The control takes place as a function of different parameters, for example from operating parameters like e.g. the number of revolutions of the internal-combustion engine 10, which is suggested by a measuring sensor 22, whose signal is given over the line 24 on the electronic control unit 18. A further input signal for the electrical control unit 18 is measured by an accelerotor pedal 26, its position in connection with the further inputs of the operating parameters of the internal-combustion engine 10 the start of injection, which specifies injecting duration and the injection pressure.

To the decrease of the losses of energy the injection pressure and thus the pressure are steered in the Common Rail 14 by a variable delivery of the high-pressure pump 16. When this control only as much fuel of the high-pressure pump 16 into the Common Rail 14 is displaced, as this is needed for adjustment the target pressure in the Common Rail 14. For this purpose a variable suction throttle 28 is intended in the intake of the high-pressure pump 16, whose flow cross-section over adjust-hurry 30 in dependence of a signal in the control line 32 is in such a manner changeable that in the Common Rail 14 a desired target pressure adjusts itself, which is measured by a pressure sensor 34 and given over the signal line 36 on the electronic control unit 18. In this way the pressure in the Common Rail 14 with small losses of energy becomes adjustable with the help of a suction throttle regulation.

With the reference symbol 38 a Vorförderpumpe is designated, which supplies the pressure fluid, i.e. the fuel by way of the suction throttle 28 of the high-pressure pump 16. The pressure downstream the Vorförderpumpe 38 is secured by a check valve 40.

The characteristic in Fig. 1 hydraulic system shown of the pressure fluid utility system exists in the mechanism emphasized with dash-dotted lines for the regulation and fast pressure relief of the Common Rail with a possibly arising emergency at the system. Such an emergency is present for example if a Injektionsnadel of an injection nozzle 121 to 12n should wedge or if from other reasons an extremely fast adjustment is necessary to a smaller load and thus the fast attitude of a smaller injection pressure. Core of this additional mechanism is pressure relief valve a named the reference symbol 50, which is activatable in arranged by the Common Rail 14 to a tank 42 prominent a line 44 and - as suggested with broken lines - across an adjusting signal under inserting a signal conversion mechanism 46 suggested with dash-dotted lines as a function of Druckund/or flow conditions in the range of the suction throttle 28. This dependence on the pressure and/or flow conditions is suggested by the broken lines 481 to 484.

In Fig. 1 pressure fluid utility system shown has the special advantage by the activating barness of the pressure relief valve 50 as a function of the pressure and/or flow conditions within the range of the suction throttle 28 the fact that at a very small technical circuiting auxiliary expenditure with the occurrence of an emergency in the CR system a fast and reliable pressure relief of the Common Rail 14 can be guaranteed, whereby the adjusting signal is usable in the line 32 for the suction throttle 28. This leads at the same time to the special advantage that operating conditions are effectively impossible, in which the high-pressure pump 16 still sufficiently with fuel is supplied, however at the same time the pressure relief valve 50 opened are, which would lead to unwanted losses of energy.

On the basis the Fig. 2 a first concrete execution form of the Aktivierungsmimik for the pressure relief valve is to be described. For the description of this arrangement similar reference symbols for components are used for the sake of simplicity, those the construction units pressure fluid of the utility system in accordance with Fig. is similar to 1, whereby a ?1? is placed in front. The representation in accordance with Fig. 2 it shows that the pressure relief valve 150 with a control member 154 of the variable suction throttle 128 over a mechanical coupling 152 stands in connection, in such a manner that the pressure relief valve 150 is adjustable by the movement of the control member 154 beyond the suction pressure control range from a closing position A into a passage position B, which leads to a fast pressure relief of the Common Rail 114. The coupling between the control member 154 and a valve body of the pressure relief valve 150 has with the fact the favourable side effect that the suction throttle 128 is regularly so far closed for the moment the Aufsteuerung of the pressure relief valve 150 that the high-pressure pump 116 cannot retard the pressure drop in the Common Rail 114 no more.

Fig. a variant of the control of the pressure relief valve shows 3. For simplification similar reference symbols are also here assigned to the description for components, which correspond to the construction units of the remark examples described before, which are upstream ?2?.

Also during this arrangement the pressure relief valve 250 connected by a mechanical coupling 252 is with the placing element 254 of the suction throttle 228. The placing element 254 of the suction throttle 228 is here formed by a valve slide of a constantly adjustable 2/2-way valve, which headed for over a control section is 256. In Fig. 3 this control section is hydraulically operated. However here that it is to be preferred, the suction throttle

228 is to be emphasized electrically, i.e. with magnet manipulation to head for. This variant is article in Fig. 4 arrangement shown.

With this variant construction units, which correspond to the elements of the execution forms described before, are provided with similar reference symbols, which however a ?3? is placed in front.

With the variant in accordance with Fig. 4 is adjust-hurry 330 by an electromagnet in an educated manner, whose armature 358 with a slidegate valve piston 354 of the adjusting throttle 328 is located in plant contact. A pivoting 360 trains a tax edge 362 for adjustment a variable throttle in cooperating with a housing bore 364, over the fuel of the Vorförderpumpe 338 as the high-pressure pump 316 is promoted. The housing is marked with 366 and takes up at the same time the pressure relief valve 350, which is formed by a poppet valve. The valve body of the pressure relief valve 350 is formed of a ball 366, which is pressed by means of a feather/spring 368 against a valve seat 370. Behind the valve seat 370 an axial drilling 371 is trained, by itself the tappet-like extension 372 of the slidegate valve piston 354 extended. In the rule position of the suction throttle 328 (as shown) affects in the Fig. 4 do not link end of the tappet 372 the ball 366 yet. The tappet has instead of its of the ball 366 an approach distance AA. With the reference symbol 373 a drilling is designated, which stands on the one hand with the tank in connection and flows directly on the other hand into the drilling 371 behind the valve seat 370.

If in the CR system a fast pressure drop should be necessary, because at the system an emergency is determined, the armature becomes 358 beyond the rule excursion in accordance with Fig. 4 to the left moves. The tax edge 362 locks the drilling 364 first completely. The slidegate valve piston 354 is continued to accelerate by the magnet 330, because the approach distance AA is not yet bridged. Only after going through this approach distance AA the tappet meets 372 the ball 366 and opens the flow path of the line 344 over the valve seat 370 and the drilling 373 to the tank suddenly, whereby a fast pressure relief of the Common Rail 314 takes place. Over the mass of the slidegate valve piston 354 and the approach distance AA the magnetic field strength can be kept small if the Ventilkugel 366 a high thrust force affects.

The execution form in accordance with Fig. it differs 5 from the remark examples described before by the fact that the pressure relief valve is 450 indirectly as a function of the flow quantity of the suction throttle 428 directly as a function of the pressure downstream the high-pressure pump 416 activatable. The suction throttle 428 is again electromagnetically operated and can be brought by spring action into a closing position, which uncouples the suction face of the high-pressure pump 416 of the Vorförderpumpe 438. The pressure relief valve 450 is switched between high-pressure pump 416 and Common Rail 414 and it reacted to the fuel river, which is caused by the displacement of the pump 416. For this purpose a cylindrical valve body 476 is axially adjustably taken up in a housing 475, whereby in an axial, gradated through-hole 477 a check valve 478 with a ball 479 is taken up. The ball 479 is subjected to a feather/spring 480 in closing position. The housing 475 has a opposite high pressure exit 482 and a discharge connection 483 in form of a drilling 485 attached to a tank line 484 beside the entrance 481. Those the high pressure exit 482 turned face of the cylindrical valve body 476 trains a tax edge 486, which changes into a chamfer 487.

The execution form in accordance with Fig. functions to 5 and 6 as follows:

As long as by means of the high-pressure pump 416 fuel is promoted to the Common Rail 414, a decrease of pressure DELTA p develops, for that the valve body 476 in in Fig at the check valve 478. 6 position shown holds, in which the Common Rail 414 is locked to the tank. There is not a flow central connection between the drilling 485 and the high pressure exit 482. The tumbler of the cylindrical valve body 476 is supported still by a difference in the surfaces, which are subjected to the pumping jerk at the entrance 481, higher around the decrease of pressure DELTA p, to high pressure in the Common Rail 414 on the one hand and.

If the suction throttle 428 in in Fig. 5 locking position shown, reach the delivery of the high-pressure pump 416 the zero and the check valve 478 are moved close. The pressure in the small volume between high-pressure pump 416 and pressure relief valve 450 sinks relatively fast by expiration of flow means, i.e. fuel over column in the pump 416, so that the pressure into the Common Rail 414 the cylindrical valve body 476 in accordance with the Fig. 5 and 6 to the left into the open position shifts, so that the pressure in the Common Rail 414 over the drilling 485 is diminished. The 490 flow paths named the reference symbols 488 and serve for maintaining a fuel stream of the Vorförderpumpe 438 over the high-pressure pump 416 for the tank line 484, in order to derive thereby from the high-pressure pump 416 warmth. In the flow path 488 a nozzle 490 is preferably inserted, with which the cooling current is controllable. Over the flow path 490 thereby additionally influence can be taken on the fast pressure drop in the high-pressure pump 416 after closing the suction throttle 428.

A further execution form of a pressure relief valve is in the Fig. 7 and 8 shown. Here the pressure relief valve 550 and/or 650 again formed by a poppet valve with a spherical valve body 566 and/or 666 is, which is printable by means of a pilot piston 590 and/or 690 against a valve seat 591 and/or 691, so that the line 544 and/or 644 to a tank line 573 and/or 673 attached to the Common Rail 514 and/or 614 is locked. The pilot piston 590 and/or 690 separates a Chamber for Employees' Welfare 591 and/or 691 from a discharge chamber 592 and/or 692. With 593 and/or 693 a nozzle is designated in the pilot pistons 590 and/or 690, over which with movement of the pilot piston fuel from the tax chamber can be displaced into the discharge chamber.

Differently with the execution forms in accordance with Fig. 7 and 8 is the kind of the control of the tax chamber 591 and/or 691.

During the arrangement in accordance with Fig. 7 the tax chamber 591 attached over a line 594 is to the suction face of the suction throttle 528 downstream the Vorförderpumpe 538. By means of a pressure relief valve 595 it is guaranteed that the pressure in the line does not exceed 594 and thus in the tax chamber 591 a certain limit value. If the Vorförderpumpe 538 is switched off with statement of an emergency at the CR system, the pressure at the exit of the Vorförderpumpe 538 and thus in the line 594 drops fast, whereupon the pressure relief valve opens 550 by shift of the pilot piston 590 fast. Over the nozzle 593 in the pilot piston 590 displaced pressure fluid, i.e. fuel over the pistons can flow off to the tank when opening the relief valve of the piston, whereby a Verkürzung of the switching time of the pressure relief valve 550 results.

The control of the tax chamber 691 during the arrangement in accordance with Fig. 8 is made by a control line 696, which is led to the suction throttle valve 628. Deviating from the remark examples described before the suction throttle valve 628 is here equipped with an additional tax edge for the supply and/or for the shut-off position of the control line 696. With the occurrence of an emergency in the CR system the placing element 654 of the suction throttle valve 628 into the closing position S is shifted, in which the control line 696 is connected by the print page of the Vorförderpumpe 638 uncoupled and with the suction face of the high-pressure pump 616. At the same time the suction face of the high-pressure pump 616 is likewise locked by the print page of the Vorförderpumpe 638. With this variant the pressure at the pilot piston 690, i.e. in the tax chamber 691 is thus lowered by switching the suction throttle valve 628, if in the CR system an emergency arises. The pressure in the tax chamber 691 drops then rapidly, so that the pressure relief valve opens 650 and makes the connection between Common Rail 614 and tank line 673.

Of course deviations from the described remark examples are possible, without leaving the basic idea of the invention. Like that it is not necessarily necessary to then activate in the inlet of the high-pressure pump with a suction throttle to work it is rather also conceivable to work with another mechanism for delivery variation as a function of the target pressure in the Common Rail and the pressure relief valve preferably as a function of condition parameters of the delivery adjusting mechanism in emergency.

The invention creates thus a pressure fluid utility system, in particular for a fuel injection system for example for a Diesel internal-combustion engine, with which to a common high pressure pipe fed by a high-pressure pump individual consumers are attached, in particular injection nozzles. The demand for pressure fluid of the injection nozzles steered as a function of operating parameters of the system, in particular the Diesel internal-combustion engine, whereby the high-pressure pump for the adjustment of the pressure in the common high pressure pipe to the pressure fluid need of the consumers a suction throttle is upstream. The throttle cross section of the suction throttle is changeable by means of a signal derived from a pressure sensor in the common high pressure pipe. In order to guarantee at very small technical circuiting expenditure if necessary a fast pressure drop in the common high pressure pipe i.e. the so-called ?Common Rail?, it is attached to a latter pressure relief valve, which is activatable preferably as a function of the pressure and/or Durch<DP N=16>strömungsverhältnissen in the range of the suction throttle.

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Result Page

The invention concerns an injection system for an internal-combustion engine in accordance with the generic term of the patent claim 1, a flow rate regulating valve and pressure control valve in accordance with the generic term of the patent claim 3 and a procedure for regulating a fuel pressure in a fuel memory in accordance with the generic term of the patent claim 7.

The regulation of the fuel pressure with a fuel memory is in particular with a Common Rail system of importance, since with a Common Rail system the maximum fuel pressure is for example at 1600 bar. Due to the high pressure it is favourable to regulate the pressure in the fuel memory with as little an energy dissipation as possible.

It is already a system well-known for the regulation of the fuel pressure in a fuel memory, with which the fuel pump pumps always too much fuel to the fuel memory and with exceeding of a given fuel pressure a pressure control valve opens. This system exhibits however a relatively low efficiency.

Further it is well-known to plan for the improvement of the efficiency a flow rate regulating valve on the inlet side of the high-pressure pump and to regulate with the flow rate regulating valve the pressure in the fuel memory. Here it is however necessary to plan additionally a pressure control valve at the fuel memory which can lower the pressure in the fuel memory fast, which for example with the transition of full load to no-load operation enterprise is necessary. Besides the pressure control valve is necessary, in order to switch the fuel memory after switching the internal-combustion engine off pressure-free.

Made of US 4.884.545 a fuel injection system is well-known, with which a fuel pump carries fuel into a fuel memory, which passes the fuel on to Einspritzventile. In the inlet to the fuel pump a flow rate regulating valve is intended, which stops the fuel stream to the fuel pump. The flow rate regulating valve is steered by a controller via an actuator. At the fuel memory a relief valve 13 is intended, which lets flow back with exceeding of a given pressure fuel from the fuel memory to the fuel tank.

In the after-published disclosure writing DE fuel injection system is described 196 12 413 A1, with which a fuel pump supplies a fuel memory with fuel, which supplies the fuel Einspritzventilen. In the inlet to the fuel pump a flow rate regulating valve is intended, which is steered by a controller. The fuel memory stands with a pressure control valve in connection, which is mechanically coupled to the flow rate regulating valve. The mechanical coupling is trained in the way that the pressure control valve is adjustable by the movement of the control member, which heads for the flow rate regulating valve from a closing position into a passage position, which leads to a fast pressure relief of the fuel memory.

The task of the invention is based to place a economical pressure control for a fuel memory ready which exhibits a high efficiency at the same time.

The task of the invention is solved by the characteristics of the requirement 1, 3 and 8. A substantial advantage of the invention is based in the fact that with only one regulating valve both the flow rate in the inlet to the high-pressure pump and the pressure in the fuel memory are regulated.

Further favourable training and improvements of the invention are indicated in the dependent requirements.

The invention is more near described in the following on the basis the figures; show:

Fig. 1 an injection system with the regulating valve according to invention,

Fig. 2 the schematic structure of the regulating valve,

Fig. 3 a further execution form of the regulating valve,

Fig. 4 a retaining pressure and a flow rate characteristic and

Fig. 5 a preferential design of the regulating valve.

Fig. 1 shows schematically the structure of an injection system, which supplies 11 fuel by way of a Vorförderpumpe 2 from a fuel tank by way of a regulating valve 10 of a high-pressure pump 1. The high-pressure pump 1 consolidates the supplied fuel and delivers under high pressure standing fuel into the fuel memory 4. The fuel memory 4 stands with Einspritzventilen 5 in connection, over which the fuel is injected into an internal-

combustion engine. To the Vorförderpumpe 2 a form regulating valve 3 is parallel switched, which stops the fuel pressure after the Vorförderpumpe 2 to a given value.

The fuel memory 4 is attached over a return pipe 26 to the regulating valve 10. The regulating valve 10 is besides to a tank line 27 attached, which is led to the fuel tank 11. At the fuel memory 4 a pressure sensor 9 is arranged, which stands over a signal line with a controller 6 in connection. Besides a number of revolutions sensor 8 and an accelerotor pedal sensor 7 are intended, which are likewise over a signal line to the controller 6 attached. The controller 6 had a data memory 28 and is connected by first control lines with the Einspritzventilen 5 and by a second control line with the regulating valve 10.

The arrangement after Fig. 1 functions as follows: The controller 6 steers the Einspritzventile 5 as a function of the number of revolutions of the internal-combustion engine and the driver desire after an appropriate program, which is put down in the data memory 28. Besides the controller 6 steers the regulating valve 10 as a function of the number of revolutions of the internal-combustion engine and the fuel pressure in the fuel memory 4 and regulates thus the fuel pressure in the fuel memory 4.

Fig. 2 shows schematically the structure of the regulating valve 10. The regulating valve 10 exhibits an actuator 21, which is designed as magnet for example. The actuator 21 stands directly with a volume closing member 20 in connection, which the fuel inlet 24, which connects comes from the Vorförderpumpe 2, with the expiration of fuel 23, which is led to the high-pressure pump 1. Besides the volume closing member 20 over a feather/spring 12 with a pressure closing member 22 stands in connection, which locks the connection between the return pipe 26 and the tank line 27 with an adjustable retaining pressure.

In the quiescent position the connecting cross section between the fuel inlet 24 and the expiration of fuel 23 closed by the volume closing member 20 is and the connection between the return pipe 26 and the tank line 27 is opened. Now if the controller 6 heads for the magnet 21, then the volume closing member 20 in the direction of the pressure closing member 22 is moved and thus the connecting cross section between the fuel inlet 24 and the expiration of fuel 23 is opened. Besides over the feather/spring 12 the pressure closing member 22 is linked up against the cross-section of the opening of the return pipe 26. Preferably the feather/spring 12 is in such a manner trained that in the quiescent position the pressure closing member 22 releases the return pipe 26 and the return pipe 26 with the tank line 27 is connected.

Fig. a further training of the regulating valve 10 shows 3, with which the effect connection between the volume closing member 20 and the pressure closing member 22 by means of a first ouple feather/spring 16 is reached and a second ouple feather/spring 17. The second ouple feather/spring 17 is linked up on a given spring action. Now if the controller 6 heads for the actuator 21, then the volume closing member 20 is moved and the flow rate, which flows to the high-pressure pump increased. Besides the pressure closing member 22 over the first ouple feather/spring 16 against the return pipe 26 is linked up. Now if the first ouple feather/spring 16 is so far squeezed together that the linked up spring action of the second ouple feather/spring 17 is reached, then first and the second ouple feather/spring 16, 17 work in series connection with further moving of the volume closing member 20.

Fig. 4 shows characteristics for the retaining pressure P for different feather/spring couplings between the volume closing member and the pressure closing member as a function of the shifting way S of the actuator 21 and as a function of the cross-section of the opening Q, which volume closing member 20 up-steers. The retaining pressure P corresponds in each case to a closing force F. deflection within the range more largely than S1 corresponds to a flow rate $Q > 0$. The rest position of the actuator 21 however preferably lies in the diagram with $s = 0$, whereby it is placed surely that in the rest position, i.e. in the unbestromten condition of the actuator 21 the fuel memory is pressure-free switched. During a deflection of the actuator 21 between 0 and S1 first the pressure closing member 22 develops a retaining pressure, before the volume closing member 20 opens the connecting cross section between the fuel inlet 24 and expiration of fuel 23 during first deflection S1.

The characteristic A corresponds to the regulating valve 10 the Fig. 2, with which only one feather/spring 12 between the volume closing member 20 is intended and the pressure closing member 22. The pressure, which is stopped by the pressure closing member 22, increases thereby linear with deflection s of the actuator 21. From Fig. 4 is recognizable that the volume closing member 20 puts at first, i.e. for deflection $s < S1$ back free travel, in which the expiration of fuel with the fuel inlet yet is not connected. The pressure closing member 22 is linked up during first deflection S1, with which the volume closing member opens the connecting cross section between the fuel inlet and the fuel derivative, with a retaining strength F0. The linear characteristic has the disadvantage that during large deflection s a large retaining strength F is developed.

The retaining pressure characteristic for the regulating valve of the Fig. 3, with a first and a second ouple feather/spring 16, 17 between the volume closing member 20 and the pressure closing member 22 is intended, is represented in the characteristic B. In the quiescent position the first ouple feather/spring 16 is relaxed and the second ouple feather/spring 17 with the help of a notice 18 and a transmission disk 19 linked up. Free travel is intended also here for the volume closing member 20, so that it opens only if the pressure closing member 22 with a retaining strength F1 is linked up already against the return pipe 26. Now if the actuator 21 is headed for and if the volume closing member 20 is expenditure-steered, then the retaining strength, with which the pressure closing member 22 is linked up, rises linear up to a second deflection S2. The linear rise corresponds to the spring rate of the first ouple feather/spring 16. Starting from second deflection S2 the first ouple feather/spring 16 is in such a manner strained that the spring action of the first ouple feather/spring 16 reaches the spring action of the linked up second ouple feather/spring 17. As soon as the tension of the first ouple feather/spring 16

exceeds the linking up strength of the second ouple feather/spring 17, the transmission disk 19 from the notice 18 separates and the second ouple feather/spring 17 is squeezed together likewise. Thus first and the second ouple feather/spring work 16, 17 starting from second deflection S2 in series connection. Therefore the linear rise of the retaining strength bends into a second, flatter linear rise starting from second deflection S2, which corresponds to a smaller spring rate. Starting from second deflection S2 the retaining strength, with which the pressure closing member 22 is linked up, per deflection unit s increases smaller than within the range between deflection s = 0 and second deflection S2, i.e. the spring rate is smaller for s > S2.

In place of first and the second ouple feather/spring 16, 17 if a feather/spring, in particular a diaphragm spring, is used which exhibits a degressive feather/spring characteristic, then a retaining strength of the pressure closing member 22 as a function of the deflection of the magnet 21 according to the characteristic C of the Fig results. 4. Due to the degressive feather/spring characteristic the retaining strength of the pressure closing member 22 increases steeply and changes in the range of second deflection S2 into an almost horizontal process with small deflections after the quiescent position s = 0. During first deflection S1 the diaphragm spring exhibits a given retaining strength F2.

By the degressive diaphragm spring or by the two ouple feathers/springs 16, 17, on the basis of the quiescent position with s = 0 a steep rise of the retaining strength is reached on the pressure closing member 22, which starting from, a second deflection given in advance S2 of the actuator 21 into a flat rise changes. The characteristic B and C are adapted to actual conditions of the fuel pressure in the fuel memory and the supplied volume stream to the fuel memory. For a COMM on Rail system already high fuel pressures are necessary with small flow rates, i.e. with small quantity of fuel, which is injected, and low engine speed. Thus the characteristics B and C offer a good efficiency for the electrical control, since unnecessarily high retaining forces are avoided with large deflections. It is favourable in such a way to insert the feather/spring with the degressive feather/spring characteristic that this is arranged in a given distance to the volume closing member or pressure closing member, because thus a structure of a counter-pressure is avoided, if the pressure in the fuel memory is diminished over the pressure closing member.

The characteristic forms B, C offer further the advantage that within the range between the quiescent position of the actuator 21 and second deflection S2 a large change of the retaining strength of the pressure stop closing member 22 is reached during at the same time small deflection of the actuator 21, whereby at the same time a small change of the deflection of the volume closing member 20 takes place and thus when small change the volume stream. In this way small flow rates can be stopped exactly. This is from Fig. 4 evidently, since the pressure retaining strength F is a function, preferably proportionally, from the control current I, with which the actuator 21 one steers. The flow rate Q is likewise preferably proportional to deflection S.

Fig. a preferential execution form of the regulating valve 10 with a feather/spring combination shows 5 according to Fig. 3. The regulating valve 10 exhibits a valve body 31, which is screwed in with a key attack 32 and a central thread 33 into a step drill of a housing 34. The housing 34 is preferably the housing of a high-pressure pump. In the housing 34 are brought in an inlet drilling 35, a sequence hole 36, a high pressure inlet drilling 38 and a high pressure sequence hole 37. To the inlet drilling 35 the fuel inlet is 24, to the sequence hole 36 is the expiration of fuel 23, to the high pressure inlet drilling 38 is the return pipe 26 and to the high pressure inlet drilling 37 is attached the tank line 27. The inlet drilling 35, the sequence hole 36 and the high pressure sequence hole 37 are designed preferably as radial ports and flow into an appropriate first ring channel to 39, second ring channel 40 and third ring channel 41. In the represented example the first ring channel 39 and the second ring channel 40 through axially shifted diameter stages in the housing 34 and in the valve body result 31. The third ring channel 41 is brought in as circulating groove into the valve body 31. In Fig. 5 represented regulating valve 10 is generally cylinder-symmetrically as the symmetry axis 71 trained.

Between the first ring channel 39 and the central thread 33 a first sealing ring is 42, between the first ring channel 39 and the second ring channel 40 is a second sealing ring 43 and between the second ring channel 40 and the third ring channel 41 is brought in a third sealing ring 44 into the valve housing 31. First man second and the third sealing ring 42, 43, 44 are designed as radially sealing O-rings.

Into the valve body 31 are brought in a first Verbindungsbohrung 55, a second Verbindungsbohrung 57 and a third Verbindungsbohrung 64 on the basis of first, second and the third ring channel 39, 40, 41. First, second and the third Verbindungsbohrung 55, 57, 64 connect first, second and the third ring channel 39, 40, 41 with a central drilling 70, which is symmetrically to the symmetry axis 71 and into longitudinal direction of the valve body 31 into the valve body 31 brought. In the central drilling 70 parallel a rule slidegate valve 53 is brought in to the symmetry axis 71, which is designed as case. Besides a closing pin 51 is intended within the rule slidegate valve 53, which is arranged symmetrically and in longitudinal direction to the symmetry axis 71. The closing pin 51 and the rule slidegate valve 53 are fit in and in longitudinal direction of the central drilling 70 adjustably arranged into the central drilling 70. The rule slidegate valve 53 exhibits a circularly rotating and annular space 54 open to the valve body 31, which is connected in the quiescent position of the rule slidegate valve 53 only with the first inlet drilling 35. If the rule slidegate valve 53 by that electromagnets 72 is moved into the working position, then by the annular space 54 the first inlet drilling 35 with the first sequence hole 36 is connected. In this way the flow rate, which is supplied to the high-pressure pump 1, is steered.

Centrically to the symmetry axis the high pressure inlet drilling 38 is brought in 71 and at the lower end the regulating valve 10. The central drilling 70 is final in the lower range by a head piece 45, which rises up at the same time with a plug into the high pressure inlet drilling 38. The head piece 45 is arranged centric to the symmetry axis 71 and exhibits centrically a pressure drain drilling 48. The pressure drain drilling 48 expand itself

In the direction of the central drilling 70 into a conical valve seat, in which a ball 50 is arranged, which is held by an admission 52 on the pressure drain drilling 48. The admission 52 forms the lower end of the closing pin 51. Between the head piece 45 and the rule slidegate valve 53 circulating an annular space 63 is trained around the closing pin 51, to which the high pressure sequence hole 37 over the third Verbindungsbohrung 64 is attached.

The cylindric continuation of the head piece 45, which rises up into the high pressure inlet 38, is surrounded by a circular back-up ring 47 and a sealing ring 46, which seal the high pressure inlet drilling 38.

By an appropriate movement of the closing pin 51 the ball 50 with an appropriate retaining strength F is subjected, so that the pressure drain drilling 48 is only released if the pressure is larger in the pressure drain drilling 48 than the retaining strength F. In this way the connection between the high pressure inlet 38 and expiration of high pressure 37 steered. The closing pin 51 with the ball 50 places a pressure closing member 22 according to the Fig. 2 and 3.

The sealing ring 46 and the back-up ring 47 offer the advantage that no axial contact pressure is needed, in order to seal the high pressure inlet drilling 38. The axial contact pressure would have to be taken up between the head piece 45 and the central thread 33 by the valve housing 31, if as seal for example a metallic flat seat or Kegelsitz or a sleeve fitting use would find.

In the upper range of the regulating valve 10 an electromagnet 72 arranged with a magnet coil 73 is and an assigned anchor guide rod 58, which are centrically to the symmetry axis 71 in a guide sleeve 80 led. The anchor guide rod 58 rises up into the central drilling 70 and is over a second ouple feather/spring 17 with the rule slidegate valve 53 in effect connection. The second ouple feather/spring 17 is linked up by a transmission socket 60, whereby the transmission socket 60 at a notice surface 62 of the rule slidegate valve 53 rests upon and the second ouple feather/spring links 17 against a stop sleeve 74 up, which is designed as end piece of the anchor guide rod 58.

The transmission socket 60 exhibits adjacent on the notice 62 one perpendicularly to the symmetry axis 71 trained guard ring 75, that with a conclusion socket 76, which represents the upper end piece of the rule slidegate valve 53, a case-like second feather/spring area 77 trains, which is limited toward electromagnets 72 by the stop sleeve 74. The stop sleeve 74 is firmly connected with the conclusion socket 76. This connection takes up the linking up strength of the second ouple feather/spring 17.

The transmission socket 60 is final on the side electromagnets with a seal plate 87. Into the transmission socket 60 rises up the closing pin 51, which locks with a transmission plate 78, which exhibits a diameter according to the diameter of the transmission socket 60. The seal plate 78 is centrically perpendicularly to the symmetry axis 71 arranged and to the symmetry axis inside mobile arranged 71 and into the transmission socket 60. Between the transmission plate 78 and the seal plate 87 a first ouple feather/spring 16 is brought in.

In the quiescent position the anchor guide rod 58 with a stop plate 79 rests against a notice 66 of the magnet housing 81. Preferably the notice 66 is for example adjustable over setting shims accordingly. The armature 82 is adjusted over a further plate 85 and the stop plate 79 on the anchor guide rod 58.

The closing pin 51 is not linked up in the quiescent position against the ball 50. Thus the pressure drain drilling 48 is opened in the quiescent position, i.e. without control for electromagnets 72. The rule slidegate valve 53 is in such a way arranged in the quiescent position that the inlet drilling 35 with the annular space 54 is connected. The sequence hole 36 is not connected however with the annular space 54.

Now if the electromagnet 72 is headed for by an output stage, then the actuator guide rod 58 in the direction of the closing pin 51 moves and transfers thereby over the second ouple feather/spring 17 and the first ouple feather/spring 16 a higher pressure retaining pressure to the ball 50. The pressure retaining strength F increases linear with deflection S of the actuator guide rod 58, as in Fig. 4 is represented.

At the same time the rule slidegate valve 53 is pushed directly of the actuator seaweeds 58 in the direction of the second Verbindungsbohrung 57. As soon as the tax edge 56 of the annular space 54 reaches the second Verbindungsbohrung 57, a flow rate flows from the inlet drilling 35 for sequence hole 36. Thus the flow rate is steered to the high-pressure pump 1. Preferably the second Verbindungsbohrung 57 exhibits the form of a rectangular lengthwise slot, which moves in direction of motion of the rule slidegate valve 53 extended, so that the cross-section of the opening is directly proportional to the way S, with which the rule slidegate valve 53. A further favourable form is a triangular cross-section of the opening. The rule slidegate valve 53 places a volume closing member 20 according to the Fig. 2 and 3.

Now if the actuator guide rod 58 up to second deflection S2 is moved, then the first ouple feather/spring 16 is so far squeezed together that the spring action of the first ouple feather/spring is 16 equal to the spring action of the second, linked up ouple feather/spring 17. Thus now during a further deflection of the actuator guide rod 58 both the first ouple feather/spring 16 and the second ouple feather/spring 17 are squeezed together.

Within the range between deflection 0 and second deflection S2 the transmission socket 60 together with the stop sleeve 74 lies and the rule slidegate valve 53. Starting from second deflection S2 the transmission socket 60 in the force equilibrium of the second ouple feather/spring 17 and the first ouple feather/spring 16 moves opposite the stop sleeve 74 of the anchor guide rod 58. In this way the increase of the retaining strength F bends 50 starting from second deflection S2 on the ball into a smaller increase per deflection unit s, as from Fig. is evident to 4.

The high pressure inlet drilling 38 is connected only then with the high pressure sequence hole 37, if a larger pressure of the high pressure inlet drilling affects the ball 50, when the ball affects 50 by the retaining strength F of the closing pin 51.

With exception of the annular space 54 all areas are connected within the regulating valve by not represented slots and drillings with the annular space 63, in order to make with a shift of the individual parts a displacement possible of the fuel.

In place of the two ouple feathers/springs 16, 17 also a degressive feather/spring between the anchor guide rod 58 and the closing pin 51 can be used. In this way a pressure retaining characteristic becomes according to the characteristic C of the Fig. 4 makes possible.

The invention concerns an injec
The regulation of the fuel press
It is already a system well-know
Further it is well-known to plan